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Friction Compensation for a Force Controlled Electric Actuator with Unknown Sinusoidal Disturbance Motion

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Abstract— **This paper presents a method of friction compensation for a linear electric motor subjected to unknown sinusoidal disturbance motions. The method uses a Coulomb friction model and applies a feedforward step signal when velocity zero crossing occurs. Velocity zero crossing estimation is achieved using an algorithm based on measured feedback velocity and force.**

Keywords – force control; friction compensation; permanent-magnet linear motor

I. INTRODUCTION

This paper aims to improve the force tracking accuracy of a permanent magnet linear electric motor subjected to sinusoidal disturbance motion. Friction compensation is an extensively studied topic in engineering control literature [1]. A common class of friction compensation technique is feedforward and feedback model based compensation. Here friction is estimated using a motion based model where input into the model is either the reference motion signal (feedforward) or a measured motion signal (feedback) [1]. Many applications of friction compensation that have been studied are for actuator positioning systems. The general lack of force sensors in typical position control systems has driven research to focus on implementing friction compensation without measuring the friction force directly. Instead friction forces are estimated using friction models based upon motion, also known as model based compensation [2]. In these systems the motion reference is known in advance so both feedforward and feedback motion based models can be used.

Classical models of friction include Coulomb and Stribeck friction models where the friction force is a simple function of velocity [1]. More complex dynamical models, such as the LuGre model incorporate rate dependent characteristics such as varying break away force and frictional lag [1]. Despite the comprehensiveness of dynamical friction models they present certain drawbacks in practical friction compensation. Complex friction models require well calibrated parameter identification and can be subject to variation due to wear and tear or changes in alignment of the rig set up. Although complex dynamic models take into account the effect of small pre-sliding displacement, usually the effects are too small to be measured by typical position sensors [2]. Another difficulty with dynamic simulation models such as the LuGre model is the requirement of an internal friction state with fast dynamics. It has been reported that the delays in the dynamic internal friction state which results from digital control was poorest at velocity reversals and this may lead to limit cycles [2]. For these reasons, although the LuGre model receives widespread attention, there has been limited practical implementation due to realization difficulties [3]. In [3] feedback friction compensation was successfully implemented for position control in a table drive mechanism with linear motors using the LuGre model and a disturbance observer. To avoid difficulties of applying dynamic friction models to typical CNC machine tools, in addition to PID control, a double pulse compensation signal was devised based on analysis of the transient friction error at velocity reversal where friction effects were greatest [4].

Other than computational issues, the practical benefits of using a friction model depend largely on the quality of motion measurements available. Velocity information obtained from the differentiation of position signals can be inaccurate due to the noise in position sensors, with the errors more significant at low velocities. Methods of filtering or estimating a more precise value of velocity usually involve a trade off in terms of signal delay and accuracy. In [4] velocity estimation was obtained by varying the sampling period of encoder counts, depending on the actuator velocity. In [5] an adaptive differential filter for velocity estimation is presented.

Similar to the objective of this study, haptic interface applications require the task of force control and friction compensation for systems subjected to an unknown disturbance motion. Haptic interface devices with force feedback are required to transmit a force towards a user who is physically controlling the device motion. Here precise force control in the presence of motion disturbance is needed for transparent force rendering [5, 6]. In these studies a force sensor is used for force feedback as it gives a more direct, reliable and accurate reading of force compared to the estimation of force from motor current. In both [2] and [7] force feedback is utilised in combination with model based friction compensation, and in [7] a comparison was conducted between different friction models. In [8] a hardware in the loop test rig for an aircraft load simulator was achieved using force feedback in combination with a feedback LuGre friction compensation model; the experimental results verified accurate force tracking performance.

When the motion disturbance is unknown, the use of reference feedforward model based friction compensation is prohibited. This leaves only the option of model based feedback friction compensation in conjunction with force feedback as the sensible approach. Initial testing of the actuator in this study showed that a proportional and integral force feedback controller is capable of compensating for friction except in velocity reversal regions. This suggests that performance levels similar to a complex dynamical model such as the LuGre model can be achieved using a simple Coulomb friction model. Research on friction compensation for a position controlled table drive in [9] demonstrated that friction characteristics are predominantly Coulomb during dynamic motion when examined using a dynamic friction model with elastic bristles.

In this paper a method of Coulomb friction compensation with force feedback control is presented. To deal with issues of motion sensor inadequacies, an algorithm which detects when velocity reversal occurs using information from both force and position sensors is utilised. The proposed method offers a simple and effective method of compensation for systems equipped with force sensors.

II. Test System and Controller Structure

The test rig consists of two actuator mounted in series with a mass and spring as shown in Fig. 1. The mass is able to move freely by sliding along the rail guide. The electric actuator is in force control whilst the hydraulic actuator is position controlled to provide a motion disturbance. The electric actuator used in the test rig is a Dunkermotoren linear motor STA38 which has a built in amplifier and current controller. The built in proportional, integral and derivative (PID) current control loop acts as the inner control loop. The force controller has the structure of a standard PI feedback controller shown in Fig. 2.

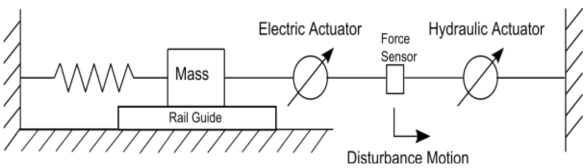

Fig. 1. Test Rig Schematic

Tests showed that outer loop proportional gain led to poor performance due to limit cycles caused by friction in the actuator and friction in the rail guide. Force tracking and elimination of steady state error is only possible by using integral gain. The controller uses a small proportional gain in combination with integral gain to limit the occurrences of limit cycles. Furthermore a feed forward reference signal is incorporated to improve the performance of the actuator since a motor's force and current are typically proportional.

Following the concept of Coulomb friction compensation the control architecture is designed to input a step command current which cancels out the Coulomb friction step change at the instance of velocity reversal. This method of Coulomb friction compensation is presented later.

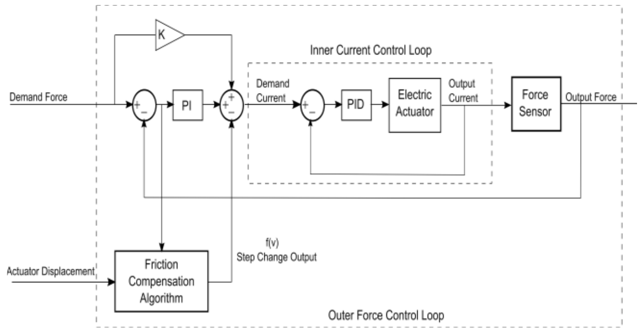

Fig. 2. Force Controller Structure.

III. Characteristic of Friction and Compensation

Fig. 3 presents the force error when subject to a 0.1-5Hz pseudo random motion input without friction compensation. This shows that at higher velocities the force tracking error magnitude is relatively small, suggesting that most viscous friction is compensated for. Compared to viscous friction, the force error caused by Coulomb friction is considerably higher as shown by the large force error spikes at zero velocity.

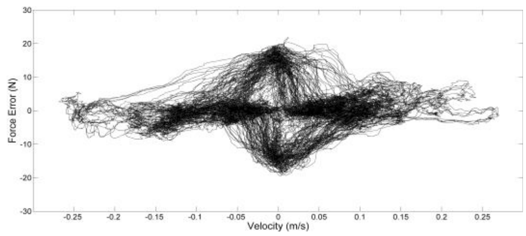

Fig. 3. Force error vs. velocity for a constant demand force and pseudo random motion.

These results suggest that major improvements in force tracking could be achieved by compensating for friction only at velocity reversal. The Coulomb friction model is chosen since it is the simplest method which captures the characteristic of friction at velocity sign changes. More complex static models and dynamic friction models are also avoided due to the difficulty of parameter identification and the requirement of very precise motion sensors [2]. This is in agreement with the finding in [9] that the non-linear friction characteristic of a table drive is expected to behave as simple Coulomb friction when under dynamic table motion. Furthermore the friction characteristics of permanent magnet linear motors are commonly modeled using classical static friction models [10, 11, 12].

The Coulomb friction model chosen for this study is described by,

$$f(v) = \begin{cases} F_c \times sign(v), & if\ v \neq 0 \\ -F_c \leq f(v) \leq F_c, & if\ v = 0 \end{cases} \tag{1}$$

Where $f(v)$ is the friction force as a function of velocity v and F_c is the Coulomb friction force.

The results of experimental friction compensation tests using a simple Coulomb friction model are shown in Fig. 4

where the control goal is constant force tracking while the actuator is subjected to a sine input motion disturbance. This demonstrates the principle of Coulomb model friction compensation assuming the zero velocity points are known in advance. Here the compensation signal is a step change of 0.15V, which is equivalent to the step change in force due to Coulomb friction.

The results in Fig. 4(a) and (b) show that the simple Coulomb model is effective at reducing the friction force spikes when compensation occurs within +/- 0.005s of the velocity reversal. Fig. 4(c) and (d) show the effect of early and late compensation applied +/- 0.01s from the velocity reversal. This results in large force error spikes and no performance improvement. If the zero velocity estimation is determined too early then the compensation signal produces a force spike, or if too late the friction spike grows before compensation occurs.

These experimental results suggest that the Coulomb friction model is effective at cancelling force spikes due to friction sign change at velocity reversal for permanent linear magnet motors, if velocity reversal times are known in advance. This is possible for motion controlled systems where velocity reversal times are determined by the reference position profile.

IV. Velocity Zero Crossing Estimation

The aim of friction compensation presented in this paper is to form a generic method for any force controlled system by utilizing a simple Coulomb friction model. Despite the simplicity of the model the previous section highlights the possible benefits and also the importance of applying the compensation signal at the precise time. The challenge therefore becomes detecting when velocity reversal has occurred, or is about to occur.

Direct velocity measurement is difficult, so often velocity is estimated from position and or acceleration measurements. The major difficulty associated with this approach is that the friction spike occurs before the obtained velocity crosses zero. This is due to both the nature of Stribeck friction but also due to lag associated with filters used to obtain a velocity signal from position data.

The method described in this paper maximizes the effectiveness of velocity reversal detection by using both motion and force information in a two stage method. The first stage is to create a window where velocity reversal is likely to occur. The second stage is then to look within the velocity window for a force profile which matches a known example friction spike which is experimentally obtained beforehand in a parameter identification process. The use of additional force information helps to improve the likelihood of successful detection and compensation.

5.1 Velocity Crossing Window

The velocity crossing window is estimated by using velocity and acceleration data at the current time step. Assuming constant acceleration, the predicted time to velocity zero crossing can be obtained from:

$$T_0 = -v/\dot{v}, \tag{1}$$

Where T_0 is the estimated time until velocity zero crossing, v and $\dot{v}$ are the velocity and acceleration of the linear electric actuator respectively.

Tests conducted on the rig for different frequencies and amplitudes of sine wave motions indicated that the start of a friction spike consistently occurred within 0.02 seconds before the zero crossing of the filtered velocity signal. Therefore a threshold of estimated velocity zero crossing time of 0.03 seconds was used to ensure that a window is activated before a force spike. This difference in time between the force spike and the velocity crossing zero varies depending on the frequency and amplitude of the sine wave motions and also the bandwidth of the velocity filter. An appropriate time till velocity zero crossing threshold T_T can be easily determined by inspection of recorded experimental data. Window activation rules are presented in Rule 1.

Rule 1 - Window Activation

Window = 1 if $(T_0 > 0)$ AND $(T_0 < T_T)$ AND $(v > 0)$

Window = -1 if $(T_0 > 0)$ AND $(T_0 < T_T)$ AND $(v < 0)$

Window = 0 if $(T_0 < 0)$ OR $(T_0 > T_T)$,

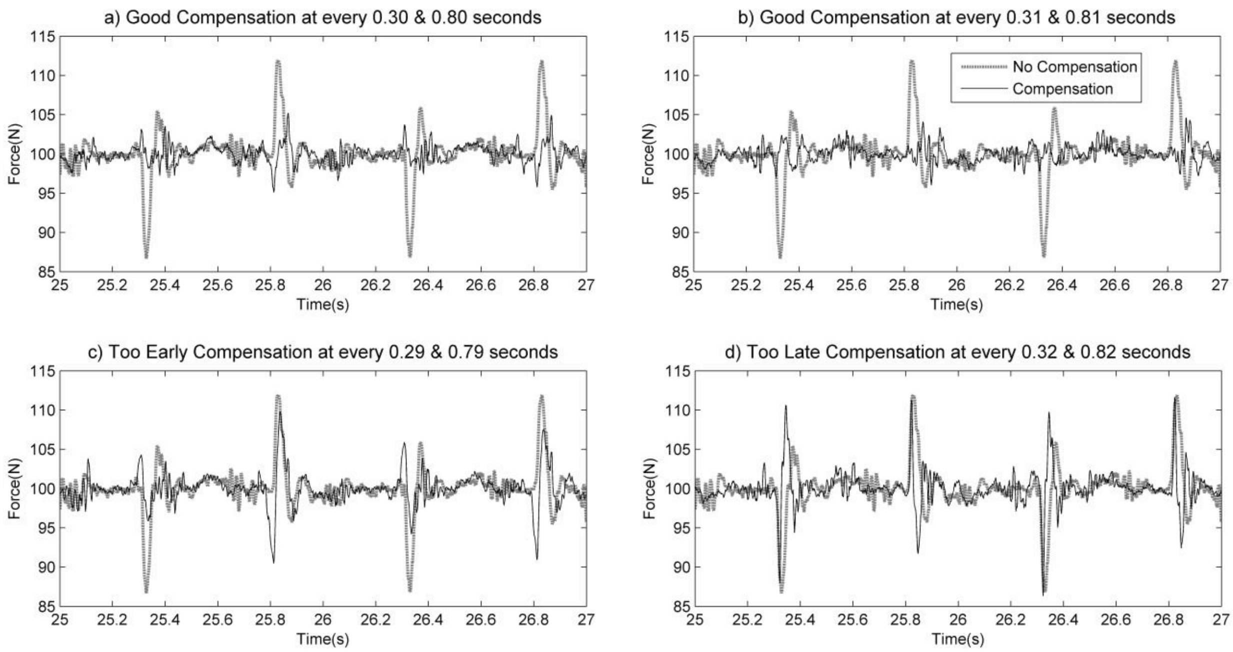

Fig. 4. Comparison of friction compensation step triggering times on 1Hz motion.

From Rule 1, when Window = 1, indicates an expected velocity zero crossing in the direction of positive to negative velocity. Conversely, Window = -1, indicates a velocity zero crossing from negative to positive. When there is no expected velocity zero crossing the Window is zero.

One implementation difficulty with this method is that if the acceleration signal is lagging the velocity signal, then in certain cases velocity reversal can occur without the window being activated. This lag would be present if acceleration is obtained by differentiating the velocity signal which would likely require filtering to remove noise.

Another solution explored here is to forward predict the velocity signal using a polynomial extrapolation which is then differentiated. Although this reduces the accuracy of the acceleration signal the improved experimental results suggest that the window is sensitive to lag of the acceleration signal. Although this method is effective in certain cases, there are also instances where because of future prediction, the obtained acceleration could be leading the velocity signal such that acceleration switches sign prior to the actual velocity zero crossing. In these cases although a window is initiated it may be cut short before a friction force spike could be detected. To alleviate this problem two estimation windows were used in combination to ensure that both cases are dealt with, one based on a short time lag acceleration signal obtained from forward prediction and the other a low pass filtered acceleration signal with a longer time lag but less noise.

Another problem that could occur is that due to noise in the velocity signal it is possible that at low velocities the signal could switch sign momentarily before the real zero crossing. In this case the window could change from active to inactive before a force spike occurs. The algorithm is adapted to deal with this issue by extending the window for a few time steps after zero velocity crossing as a preemptive action that the velocity zero crossing may be due to noise.

5.2 Force Spike Detection

The friction force spike detection algorithm is only active when the velocity zero crossing window is on. Since the force spikes are a result of the interaction between the controller, sensors, friction, motion and force trajectories, the profile of the force error spike is not predictable. However there are still generalities that can be established, such as a minimum gradient of the rise, and a force error threshold which can be used to distinguish the friction spike from those caused by other errors. The aim is to detect the initial rise of the force error spike so that compensation is applied as quickly as possible for maximum effectiveness. Rule 2 demonstrates the concept and application of the force spike detection algorithm.

Rule 2 - Friction Force Spike Detection Algorithm

Condition 1: Velocity zero crossing window is on, the direction of the expected crossing determines the direction of the expected gradient change and force error.

Condition 2: The absolute force error (F_E) of the current time step (i) must be equal or greater than the force threshold (F_T).

Condition 3: For a specified number of previous time steps (N), define a minimum absolute gradient threshold (G_T) between the force error for each subsequent time step.

Rule 2 expressed logically,

Positive Force Spike Detect = True,

if (Window = 1) AND ($F_E(i) > F_T$) AND ($| F_E(i-n+1) - F_E(i-n) | > G_T(n)$) for n = 1:N

Negative Force Spike Detect = True,

if (Window = -1) AND ($F_E(i) < F_T$) AND ($| F_E(i-n+1) - F_E(i-n) | < G_T(n)$) for n = 1:N

In regards to Rule 2, the specific thresholds and number of samples chosen for Condition 2 and Condition 3 were determined by examination of a friction force spike obtained experimentally. An example for this process is illustrated in Fig. 5.

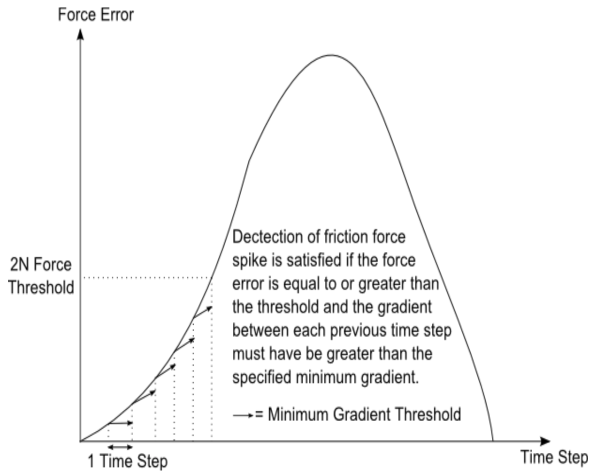

Fig. 5. Example diagram of force spike profile envelope for Condition 2 and 3 used in this study for a force threshold of 2N..

Condition 1 defines the expected direction of the force error and gradient change which is determined by the direction of the expected zero velocity crossing. This constrains the direction of force error and gradient in Condition 2 and 3.

Condition 2 assigns a minimum force error threshold so that the friction force spike is detected only when a substantial force error has occurred. Choosing a low error threshold allows earlier detection of friction force spikes but also increases the likelihood of false detections due to noise in force readings. To ensure an improvement in force tracking performance the error threshold should be chosen so that it is within the bounds of force tracking error not caused by friction. For the example in Fig. 5, the force error threshold is chosen at 2N, this is below the no motion force tracking error which is between +/- 6N.

Condition 3 introduces a gradient threshold which is also obtained from observation of the friction force spike. After choosing a force error threshold, from the friction force spike diagram, the number of samples can be chosen so that gradient change from the beginning of the force error rise up to the

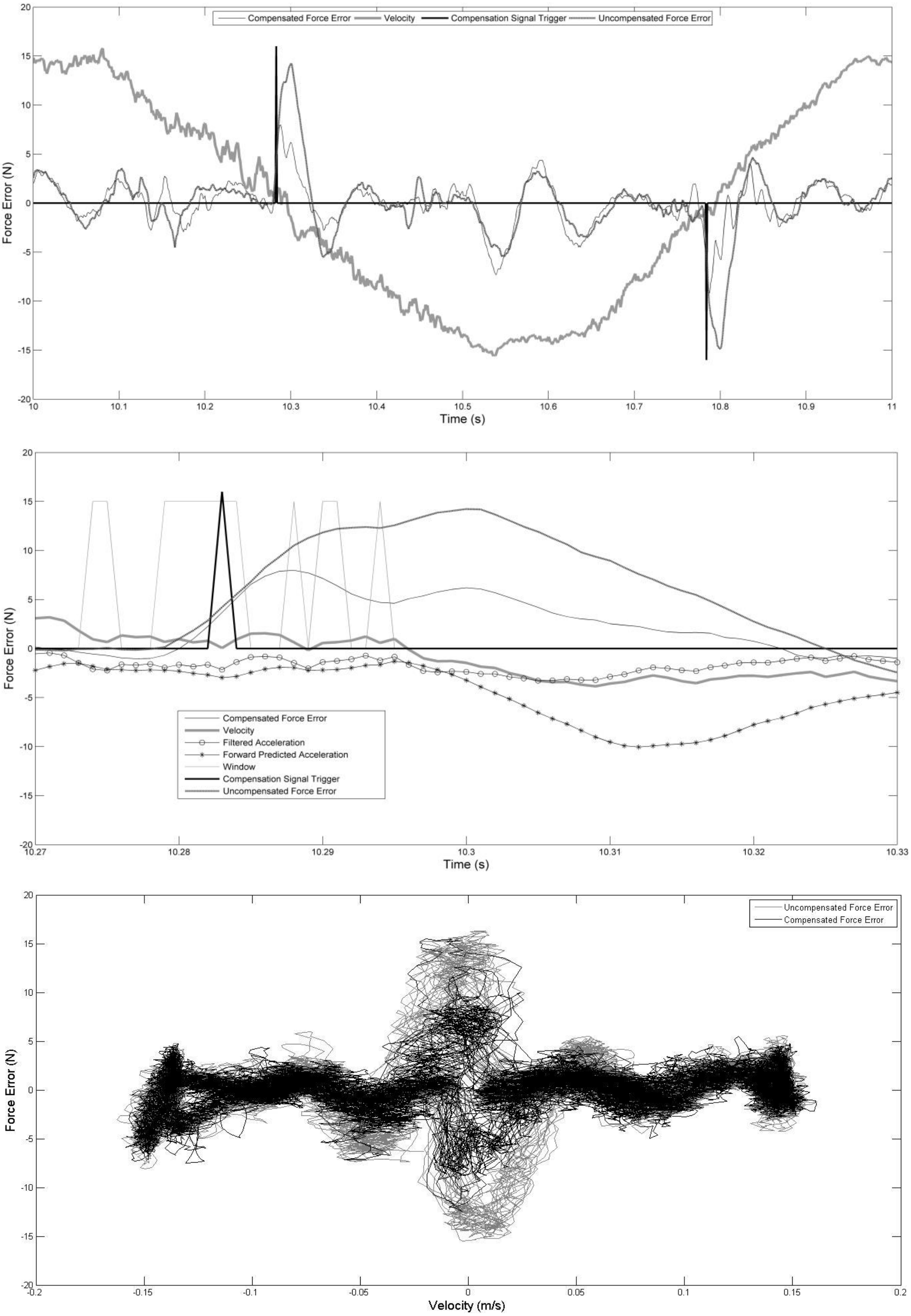

Fig. 6. Results for 1Hz and 20mm amplitude sine wave. From top to bottom. a) Time Response, b) Exploded View Time Response, c) Force Error vs. Velocity Relationship

threshold is constrained. This is simply creating a gradient change envelope constraining the shape of the friction force error spike rise. Force error due to noise tends to be more oscillatory without such consecutive force changes in the same direction. The force profile envelope condition could be further defined to include not just the gradient but the complete force versus time profile of the friction force error spike rise. For this study, experimental results showed that good detection performance was achieved using a simple gradient envelope.

The choice of the gradient envelope is illustrated by an example in Fig. 5; five gradient samples are chosen to cover the majority of the force error rise, the initial sample has a minimum gradient of zero and the subsequent samples a minimum gradient of 0.2. The minimum gradient threshold is chosen to be substantially lower than the gradient of the actual friction force spike, this is to improve robustness and demonstrate that the capability of the algorithm does not require tight force error gradient envelope constraint.

V. EXPERIMENTAL RESULTS

The friction compensation algorithm was tested with constant force tracking and single frequency and amplitude sine wave motions. The algorithm is capable of compensating friction for sine wave motions of 1-5Hz. Fig. 6 present the results for the 1Hz case. The velocities and accelerations on the plot are scaled for clarity. The force threshold was set to 2N to demonstrate the ability of the algorithm to correct for the friction spike early on in the force spike rise. The test was conducted for 25 seconds and the algorithm successfully detected and reduced all friction spikes. Table 1 compares the force tracking performance between compensation and no compensation for 1, 3 and 5Hz sine motion.

TABLE I. FORCE TRACKING PERFORMANCE COMPARISON

	RMS Force Tracking Error (N)		***RMS Improvement***	***Peak Force Error Reduction***
	No Compensation	***Compensation***		
1Hz	3.551	2.335	34.0%	44%
2Hz	6.597	3.993	39.4%	33%
3Hz	7.812	4.476	42.7%	25%

VI. CONCLUSION

The friction compensation technique presented in this paper demonstrated the ability to improve the tracking performance of force control systems. The biggest advantage of this technique is that it could be implemented on force controlled systems system without the need for modeling of the actuator, rig setup, or the friction except for a simple Coulomb model. Friction parameter identification, which can be troublesome for complex models is avoided. This algorithm only requires the determination of the integrator level change caused by Coulomb friction, the determination of (T_T) by observing time between filtered velocity zero crossing and the initial friction force spike rise and the determination of F_T, G_T and N from observation of the friction force error spike. A force sensor and position sensor for the actuator stroke are the only instruments required. This means that this technique could be implemented with relative ease on any force control system, to achieve significant improvement in force tracking.

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